NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS

TECHNICAL NOTE

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PRELIMINARY INVESTIGATION OF A GAS TURBINE WITH SILLIMANITE CERAMIC ROTOR BLADES

By Frederick J. Hartwig, Bob W. Sheflin
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SUMMARY

A gas turbine with rotor blades of a sillimanite-base ceramic material was designed, constructed, and operated to determine the practicability of ceramics for gas-turbine blading. Two blade types were used. The second design, developed for increased strength to correct the cause of failure of the first design, was simplified to facilitate fabrication but its aerodynamic properties were slightly compromised. The unit was operated at temperatures up to 1725° F and at speeds up to 10,000 rpm; however, the operating conditions were not typical of gas-turbine operation owing to lower-than-normal speed, power extraction, and pressure ratio across the turbine. The blades failed at a relatively low centrifugal tensile stress. The design criteria were somewhat uncertain; no data were available on this type of turbine because the present investigation was the first of its kind. Subsequent theoretical investigations have shown the existence of stress concentrations caused by deflections that satisfactorily explain the present failures and offer more practical means of adapting brittle materials to rotating turbine components. Investigations have shown that advantageous applications of ceramics to stressed components of gas turbines can be made with probable success.

INTRODUCTION

The low density, high melting point, and potentially low cost of ceramic materials indicate their possibilities for high-temperature turbine application. Patents indicate that ceramic materials were considered for turbine components in Germany before World War II (references 1 to 4) and the lack of certain alloying metals in Germany during the war intensified the search (reference 5). United States Army intelligence reports (references 5 and 6) covering surveys of this work indicated generally unsatisfactory results although some aspects were promising. Most ceramic materials in their present state of development have a low value of tensile strength and are quite brittle. Their strength-to-density ratio, especially at the higher temperatures, is sufficiently high, however, to warrant their investigation for use as aerodynamic elements of gas turbines, particularly for use in expendable missiles. High tensile strength ceramics in the present state of development generally have poor heat-shock properties.

The melting points of the ceramic materials considered range from 2300° to 7500° F (references 7 and 8); whereas the melting points of carbon steels are about 2700° F, those of most stainless steels about 2550° F, and those of two currently representative turbine-blade alloys about 2350° F.

The power output of gas-turbine power plants improves with increasing turbine-inlet gas temperature. The fuel consumption of turbojet power plants up to gas temperatures higher than those that can be used at present and the fuel consumption of turbine-propeller power plants improve with increasing turbine-inlet gas temperatures. Consequently, the desirability of higher operating temperatures is apparent. The strength of available materials at high temperatures imposes definite limitations on turbine-inlet gas temperatures that can be used without seriously reducing the operating life of the turbine. Allowable inlet gas temperature of present conventional turbines is limited to approximately 1500° F, but conventional fuels could provide inlet gas temperatures exceeding 3500° F. In order to maintain the trend of increasing turbine-inlet gas temperatures, cooling must be provided and cycle efficiencies must compensate for the losses chargeable to cooling, or materials with higher melting points than those now used must be sought.

On the basis of this information, an investigation of ceramic-blade turbines was initiated at the NACA Cleveland laboratory. Two blade forms were designed and made of the best ceramic material available at the beginning of the investigation. The design criteria were somewhat uncertain. No data were available on this type of turbine, particularly the blade-mounting details, because the present investigation was the first of its kind. The second blade design of increased strength was developed to correct the cause of failure of the first-design blades; the second design was simplified to facilitate fabrication, which resulted in a slightly less desirable blade aerodynamically. The operational results of these blades in a turbine are presented with a description of the blade designs, dimensional inspection, and heat-shock tests. No determinations of aerodynamic performance and the effects of constructional compromises on this performance were made because operating speeds were below the design value.

TURBINE WITH FIRST-DESIGN BLADES

Turbine Rotor Design

In turbine blades, the primary stress is due to centrifugal force and, inasmuch as centrifugal force, and therefore tensile stress, is directly proportional to density, the strength-to-density ratio of a blade material is more important than its absolute strength. A survey of the available ceramic materials indicated that sillimanite ($Al_2O_3 \cdot SiO_2$) had the best strength-to-density ratio. In specimen tensile studies (reference 9), the stress-to-rupture strength for a life of several hours was approximately 7000 pounds per square inch at 1800° F.

Turbine-design criteria. - The ratio of blade height to wheel diameter of a small commercial turbosupercharger turbine that was available was used with the density of sillimanite (0.101 lb/cu in.) and the maximum allowable blade stress (7000 lb/sq in.) to compute the pitch-line blade speed of the turbine. The maximum pitch-line speed was found to be 1000 feet per second corresponding to a speed of 19,000 rpm. Because this speed was well within the normal turbine operating range, all possible parts from the turbosupercharger were used in developing the ceramic-blade turbine.

First-design blades. - For the first blade design (fig. 1) an unshrouded, untapered impulse blade was chosen instead of a reaction blade because of its ruggedness and simplicity. The base of the blade was designed for a compressive rather than a bending loading because the compressive strength of most ceramic materials is more than 10 times greater than the tensile or bending strength (reference 10). The blades were dry-molded, machined to size, and sintered at approximately 3100° F. The sintering operation permitted tolerances of ±0.002 inch. Closer tolerances were needed only in the thickness of the root section where 86 blades fit together to form a complete circle; these faces were ground to a tolerance of ±0.0005 inch.

Upon receipt from the manufacturer, the blades were inspected as follows:

1. The surface was visually examined for external flaws.

2. X-rays were taken to detect internal flaws.

3. The blades were weighed.

4. The root angle was calculated from a thickness determination made at two points on the radial center line of the blade.

5. The width of the root section was measured with micrometers and the blades were checked on an optical comparator against a 10-times-size templet. From this inspection any deviation in the angular relation of the blade with the root could be checked as well as the correct location and shape of the contour where the blades fitted against the disks.

Approximately 5 percent of the blades were rejected because of internal flaws and 2 or 3 percent were rejected for other reasons.

Turbine-rotor disks. - A pair of disks designed to support the blades by means of a lip overhanging the blade bases were made of a nickel-base high-temperature alloy. The blade-clamping surfaces of the disks were contoured to provide space for a gasket between the blade bases and the disks. Provision was made for the disks to be clamped together against a shoulder on the shaft by means of a cap nut on the end of the shaft.

Compressive loadings of various thicknesses of asbestos gasket material were made to determine radial movement of the blades under centrifugal loading, to evaluate the effect of gasket thickness on stress concentrations in the blade base due to clamping and supporting forces, and to determine the most desirable precompression of the gaskets when assembling the blades in the disks. A single blade was forced against a heated disk section in a compression testing machine to simulate the action of centrifugal force pressing the blades against the disks while the unit was operating. The results indicated that the gasket would remain in satisfactory condition if its temperature was kept below 1200° F. The results also indicated that a nominal gasket thickness of at least 0.020 inch was necessary to prevent failure of the blade base due to stress concentrations at stresses comparable to a blade pitch-line speed of 1000 feet per second. The gasket also proved to be a good thermal barrier between the disks and the blades.

Turbine-rotor assembly. - The blades were assembled in a special wooden jig (similar to that shown in fig. 2), which held them by their tips in the correct relation to each other and at the correct radius for assembly with the disks. A gasket of high-grade commercial woven asbestos cloth with a nominal thickness of 0.062 inch (as purchased) was then fastened to the blade bases with a quick-drying cement. The disks were assembled on the shaft with the blades in place and the cap nut was pulled down. In order to make certain that the gasket was compressed to the desired thickness, previously determined measurements across the disks were checked.

Turbine Setup

Turbine. - The turbine disks and shaft were designed to use as basic equipment the inlet collector, the nozzle ring, the bearings, the lubrication system, and the main housing of a small commercial turbosupercharger. The wheel and shaft assembly with the blades clamped in place (fig. 3) was balanced on a dynamic balancing machine and then assembled with the turbosupercharger unit. The entire unit was then mounted on a specially constructed table designed to facilitate and maintain alinement between the turbine and a water brake. The turbine was connected to the water brake through a high-speed coupling.

In order to prevent a pressure difference across the main housing and a resulting flow of air or gas through the seals and bearings of the turbine, a seal pressure chamber was added to the coupling end of the turbine (fig. 4) and a pressure connection was made to the exhaust duct. The disks were cooled by passing air through the hollow turbine shaft, between the disks and the air-guide plates, and out through the holes in the cap nut. (See fig. 4.) The areas of the air passages were so proportioned that the maximum velocities and the maximum rates of heat transfer occurred at the disks just below the base of the blades. Heat-transfer calculations indicate that the cooling air will be sufficient to keep the rotor temperature below 1200° F at turbine design speed with an inlet gas temperature of 2000° F and a choking mass flow through the nozzle of 70 pounds per minute. Cooling air was supplied to the chamber surrounding the slotted end of the hollow turbine shaft.

Hot-gas system. - The induction system consisted of an orifice tank for measurement of the air flow, an air filter, a combustion chamber, and a straight section of pipe to allow thorough mixing of the products of combustion before they enter the turbine. The combustion chamber was designed for low internal-flow velocities and was suitable for producing temperatures from 150° to 2000° F at air flows from 2 to 200 pounds per minute. The low starting temperature was incorporated in the burner design to minimize heat shock to the blades during starting. The turbine exhausted into the laboratory low-pressure exhaust system through an annular discharge duct.

Instrumentation. - A minimum amount of instrumentation was used because the primary purpose of this investigation was to determine if a turbine with ceramic blades could be run at the same or higher inlet gas temperatures than a metal-blade turbine. Turbine-inlet pressure was measured by means of a mercury manometer connected to a static-pressure ring installed on the inlet duct 12 inches ahead of the turbine-inlet collector. Turbine-exhaust pressure was measured by means of a mercury manometer connected to a static-pressure ring installed on the exhaust duct 36 inches downstream of the turbine.

The inlet-air flow was measured with a micromanometer connected across a 10-inch plate orifice in an orifice tank. Fuel flow was measured with rotameters.

Triple-shielded thermocouples installed at the entrance to the turbine-inlet collector measured the gas temperature.

Turbine speed was measured with an electric tachometer and checked with a chronometric tachometer.

Vibration was indicated by a piezoelectric crystal pickup and amplifier. The crystal was horizontally mounted on the turbine-support table and was enclosed by a water-cooled jacket. The calibration of the instrument was not used after the initial vibration check was made but comparative readings were made to indicate major changes in amplitude of vibration so critical speed ranges could be avoided and excessive unbalancing of the rotor due to shifting of the blades or blade breakage could be detected.

Procedure with First-Design Blades

Nozzle-temperature variation. - The temperature variation that existed between nozzles or at different positions in any one nozzle was measured to evaluate the danger to the blades from heat shock. Thermocouples were installed in every fourth nozzle passage in the nozzle diaphragm and three thermocouples were placed in different radial positions in two of the nozzles. The setup was then operated without a turbine wheel at temperatures varying from 700 to 1650° F. The greatest gas-temperature variations between nozzles were 25° and 125° F at inlet gas temperatures of 750° and 1600° F, respectively. The greatest radial variations in a single nozzle were 20° and 60° F at inlet gas temperatures of 750° and 1600° F, respectively. These small temperature differences would not seriously affect the blades because the depth of penetration of the temperature fluctuations into the blade would be very small at any reasonable operating speed.

Heat-shock procedure. - In order to determine the ability of the blades to withstand heat shock, a rig (fig. 5) was set up for tests on the actual blade because heat-shock characteristics are a function of the shape of the body. A single blade was mounted in a simulated nozzle-box section with a quartz window installed for observation. A stream of gas was directed across the blade at sonic velocity and the temperature of the gas was varied from approximately 400° F to a maximum of 1500° F by changing the postion of the butterfly valve (fig. 5), which interchanged the paths of the hot-gas and cold-gas streams. A duplicate nozzle was installed in the waste pipe to introduce the same amount of flow resistance in both flow paths thus minimizing changes in the operating conditions of the burner during temperature cycles. By varying the rate of position change of the butterfly valve, the rate of temperature change with time could be varied. A high-speed recording potentiometer reading from a high-response shielded thermocouple located just ahead of the nozzle made it possible to keep a record of the change of temperature of the gas stream. The blades successfully withstood heating cycles with an 1100° F temperature change at the rate of 24,000° F per minute, which was the limit of the setup. During the cooling cycle, two out of three blades broke with the same degree of temperature change and the same rate of temperature change. The blades withstood repeated cooling cycles with an 1100° F temperature change at the rate of 20,000° F per minute.

Turbine procedure. - In starting the turbine, the exhaust pressure was set to give a low pressure ratio (1.15) across the turbine and the burner was started with a fuel flow just sufficient to give an inlet gas temperature of 150° F. The temperature was slowly increased to 500° F and the exhaust pressure decreased to give the pressure ratio desired. The speed of the turbine was gradually brought up to 1000 rpm by releasing a band brake on the water brake, which until this point in the starting procedure had prevented the turbine from rotating.

Even though the preliminary data indicated that the material would withstand considerable heat shock, the precautions involved in this method of starting were always observed because the single-blade heat-shock investigation did not exactly duplicate the turbine conditions with respect to method of support and centrifugal stress combined with thermal stress. This method was used to minimize any heat shock to the blades due to uneven heating of the nozzle box and to protect the blades as much as possible in case there should be a backfire or other difficulty in starting the burner. Throughout the investigation, the turbine speed was controlled by adjusting the water brake and the pressure ratio across the turbine; the inlet gas temperature was controlled by adjusting the fuel flow.

Calculations indicated that the first critical speed would occur at about 7000 rpm and a plot of vibration reading against speed indicated that the maximum amplitude occurred at slightly over

7000 rpm. The unit was therefore run through the speed range of 6000 to 9000 rpm as rapidly as possible and readings were continuously taken on the vibration meter.

After completion of a run, the speed was decreased to approximately 1000 rpm by loading the water brake, and the fuel flow was gradually reduced until the flame went out at an inlet gas temperature of approximately 125° F. The pressure ratio was then reduced to 1.15 and the band brake used to bring the turbine to a complete stop.

Although the burner was designed for temperatures up to 2000° F, stresses on the ducting and other parts of the setup made operation at inlet gas temperatures above 1800° F undesirable.

Results with First-Design Blades

The turbine was initially operated at low rotative speeds and the inlet gas temperature was increased from 200° to 1600° F and then reduced to 500° F over a period of $1\frac{1}{2}$ hours. (See table I.) At an inlet gas temperature of 500° F, the speed was gradually increased to 10,000 rpm at which speed with a total running time of $1\frac{1}{2}$ hours the blades failed. The unit after failure is shown in figure 6.

The setup and the turbine were carefully examined to determine the cause of failure but no positive reason could be found other than the effect of stress concentrations at the 1/32-inch-radius fillet between the aerodynamic part of the blade and its base. Most of the blades failed at this fillet with what appears to be a simple tension break. (See fig. 6.) The blades broken above this point were probably fractured by flying fragments after the rupture of the first blade, or blades, at the root.

TURBINE WITH SECOND-DESIGN BLADES

Blades of Second Design

The second-design blades differed from the first because of the necessity for reducing stress concentrations in the blade. The blade base was increased in thickness perpendicular to the chord to eliminate the tongue and groove in the root section (figs. 1 and 7) and the radius of the fillets at the most critical section, that is, at the bottom of the blade section, was increased from 1/32 to 1/8 inch to reduce stress concentrations. The complete wheel had 58 blades. A gasket with a nominal thickness of 0.025 inch was used between the blades and the disks instead of the 0.062-inch gasket used with the first-design blades in order to reduce the amount of radial movement of the blades under the action of centrifugal force. In order that the same disks could be used, the blade size was increased by the amount the gasket thickness was reduced. The radial movement of the blades allowed by the compression of the supporting gasket increases the circumferential clearance between the blades. A 0.020-inch asbestos gasket preloaded to about 500 pounds per square inch was therefore placed between the blade bases to eliminate any further movement of the blades relative to each other. One strip of gasket material was run around the root section of alternate blades as shown in figure 8.

Data reported in reference 1 showed that heat treatment of the material for 1/2 hour or more at 1800° F considerably increased the strength in the low-and medium-temperature range. The first-design blades were not heat-treated because the favorable effect on tensile strength, particularly in the 1000° to 1400° F range, was not known at the time the first turbine was run. The second-design blades were heat-treated by placing them in a furnace, slowly increasing the temperature to 1800° F, maintaining this temperature for 1 hour, and then decreasing to room temperature again over a period of 3 hours. The blades were inspected in the same manner as the first set and again approximately 5 percent of the blades were rejected because of internal flaws and approximately 3 percent for other reasons.

Turbine-Rotor Assembly and Setup

The blades were assembled into a ring (fig. 9) in a special wooden jig (fig. 2). The remainder of the assembly, balancing, and alinement procedure was the same as used for the blades of the first design and no changes were made in the setup. Figure 10 shows the rotor assembly.

Test Procedure with Second-Design Blades

The heat-shock tests showed the characteristics of the second-design blades to be the same as those of the first-design blades.

The operating procedure for the turbine with the second-design blades was the same as for the first-design blades except that at an inlet gas temperature of 1000° F, it became apparent that some misalinement of the water brake and turbine existed due to pressure exerted on the nozzle box by the exhaust duct as the duct became heated. (See remarks in table I.) For subsequent runs, the turbine was held stationary and the gas temperature increased to 1200° F while alinement adjustments were made, after which the gas temperature was decreased to 500° F and the band brake released.

Results with Second-Design Blades

The turbine was run for a total time of slightly more than 38 hours at speeds up to 8700 rpm and at temperatures up to 1725° F. Included in the log of operation were 30 starts and 36 temperature cycles. In attempting to keep vibration as low as possible, the turbine was balanced so well that no apparent critical speed due to unbalance of the turbine wheel was encountered at any point. Numerous relatively small changes in vibration occurred at varying speeds, sometimes as little as 200 rpm apart. These changes were probably due to secondary vibrations in the multiple-disk water brake.

After each day's running, a section of the exhaust ducting was removed, the blades were examined for looseness, and the gasket material was checked for evidence of deterioration. After the short run at 1725° F (table I, 23 accumulated hr), the unit was completely dismantled to examine the blades and the gaskets. The gaskets were found to be in excellent condition with no sign of deterioration beyond some blackening caused by oil that had seeped into them. Several of the blades were found to have minute chips off the corners near the bottom of the base, probably the result of thermal stresses arising from the sharp temperature variations where the wheel was close to the blades and the cooling air flowed by the blades. These blades were then replaced, the unit reassembled, and the running continued.

After a total running time of 38 hours (including 5 hr during which the blades were subject to hot-gas flow but the turbine was stationary), a run was to be made at 8500 rpm and 1500° F. The turbine was brought up to a speed of 8100 rpm and a temperature of 1500° F; the speed was gradually advanced to 8700 rpm at which point it fell off to about 6700 rpm owing to an instability in the water brake. The turbine was again brought up to 8300 rpm; after about 5 minutes at this speed, the turbine failed. Figure 11 shows the unit after the failure. A careful examination showed no mechanical reason for the failure. Several blades failed inside the wheel rather than at the base of the aerodynamic section, as occurred in the failure of the first-design blades.

Because of the blade failures at the supporting shoulders, a more detailed analysis of the stresses in the blade base than had been made during the design, was begun. The calculations of compressive stress on the supporting shoulder showed the existence of stress peaks coinciding with the intersection of the line of failure with the supporting shoulders of the blade. These stress peaks could be eliminated by redesign of the blades or supporting disks. Calculations of the stress distribution on these shoulders due to centrifugal loading were made for blade-shoulder and disk-rim contours that were parallel before centrifugal loading when known load-deflection characteristics of the gasket material were used and zero axial deflection of the disks was assumed. This calculation resulted in a computed peak compressive stress of 100,000 pounds per square inch at 10,000 rpm. For a computed axial deflection of 0.030 inch for each disk, the peak compressive stress increased at least 185,000 pounds per square inch (based on extrapolated gasket-load compression data) at 10,000 rpm with only a minute change in its location. The possibility of a more accurate analysis of the stresses in a ceramic blade is questionable at this time owing to the many unknown factors affecting the calculations such as the actual axial deflection of the turbine disks and complete physical-property data on ceramics.

General Considerations

The conditions under which the ceramic gas turbine was operated were not typical of normal gas-turbine operation because only enough power was extracted from the turbine to keep the water brake in a stable range. The angle of flow into the blades therefore varied considerably from the blade angle.

The point of failure of a ceramic-blade turbine was shifted from the junction of the blade section and its base by reduction of the stress concentration at this point. The second failure of the blade coincides with a high calculated stress peak in the area of attachment due to the radial movement of the blades and the axial deflection of the disks.

Although the turbine was not operated extensively at inlet gas temperatures above 1650° F, the results described in reference 9 indicate that the blade material should operate satisfactorily up to 1800° F, corresponding to an estimated inlet gas temperature of approximately 2000° F for this turbine.

CONCLUSIONS

The results of this preliminary investigation and the known physical properties of ceramics indicate that ceramic-blade turbines can be operated at high inlet gas temperatures, 1800° to 2000° F, and moderate speeds for short life. With greater care in design, it is believed that possibilities for increasing both load and life warrant continuation of research on turbines of ceramic material.

Flight Propulsion Research Laboratory,
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Cleveland, Ohio, May 23, 1947.

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TABLE I - RECORD OF RUNNING TIME OF CERAMIC-BLADE GAS TURBINE

Run	Time (hr)	Total running time (hr)	Inlet gas temperature (°F)	Turbine speed (rpm)	Remarks
Unit With First-Design Blades					
1	1/4	1/4	200	0	Unit held stationary
2	1/4	1/2	500	20	Speed controlled by band brake
3	1/2	1	500-1600	100	Speed controlled by band brake
4	1/2	$1\frac{1}{2}$	500	1000-10,000	Blades broke at approximately 10,000 rpm
Unit With Second-Design Blades					
1	1	1	500	1500	
2	2	3	600	1500	
3	2	5	700	5400	Instability in water brake allowed turbine speed to reach 7500 rpm for few minutes during run
4	2	7	1000	5000	Instability in water brake allowed turbine speed to reach 8500 rpm for few minutes during run. Coupling vibration made it necessary to shut down
5	1	8	500	1200-2400	Shut down due to high vibration pickup reading
6	1/2	$8\frac{1}{2}$	500	1000-3100	Shut down due to high vibration pickup reading
7	1/2	9	80-750	0	Variation in alinement with changes in gas temperature investigated
8	1	10	90-1000	0	
9	2	12	500-1200	1500	

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TABLE I - RECORD OF RUNNING TIME OF CERAMIC-BLADE

GAS TURBINE (continued)

Run	Time (hr)	Total running time (hr)	Inlet gas temperature (°F)	Turbine speed (rpm)	Remarks
Unit With Second-Design Blades					
10	1	13	1200	0-4300	Unit alined at inlet gas temperature of 1200° F
11	2	15	1300	5700	
12	2	17	1500	6300	
13	1/2	$17\frac{1}{2}$	1250	0	Checked alinement at inlet gas temperature of 1250° F
	1	$18\frac{1}{2}$	1250	5400	
14	3	$21\frac{1}{2}$	1600	5700	
15	1/2	22	1300	0	Checked alinement at 1300° F
16	1	23	1600-1650	5300	Vibration pickup reading, higher than normal
17			1725	5400	Radiation from nozzle box caused small fire necessitating shut down after only few minutes of operation
Completely Rebuilt Unit With Second-Design Blades					
18	$1\frac{1}{2}$	$24\frac{1}{2}$	1200	0	Unit alined at inlet gas temperature of 1200° F
	1/2	25	1200	1400	
19	1/4	$25\frac{1}{4}$	1200	0-1500	
	3/4	26	1200	4500	
20	1	27	1400	4500	

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TABLE I - RECORD OF RUNNING TIME OF CERAMIC-BLADE GAS TURBINE (concluded)

	Completely Rebuilt Unit With Second-Design Blades				
Run	Time (hr)	Total running time (hr)	Inlet gas temperature (°F)	Turbine speed (rpm)	Remarks
21	3	30	1600-1650	4500-6000	Tried speeds up to 6000 rpm but large vibration indicated at all speeds over 5000 rpm
22	1	31	80-500	0	Checked movement of nozzle box with changes in exhaust-duct temperature
23	1/2	$31\frac{1}{2}$	500-1200	1500	
24	2	$33\frac{1}{2}$	1200	500	
25	1	$34\frac{1}{2}$	1200	7500	
26	1/2	35	500	1200	
27	1	36	1200	4200-7300	
28	1	37	1400	7000	
29	1	38	1500	7000	
30	10 minutes		1500	8100-8700	Unit failed at 8300 rpm

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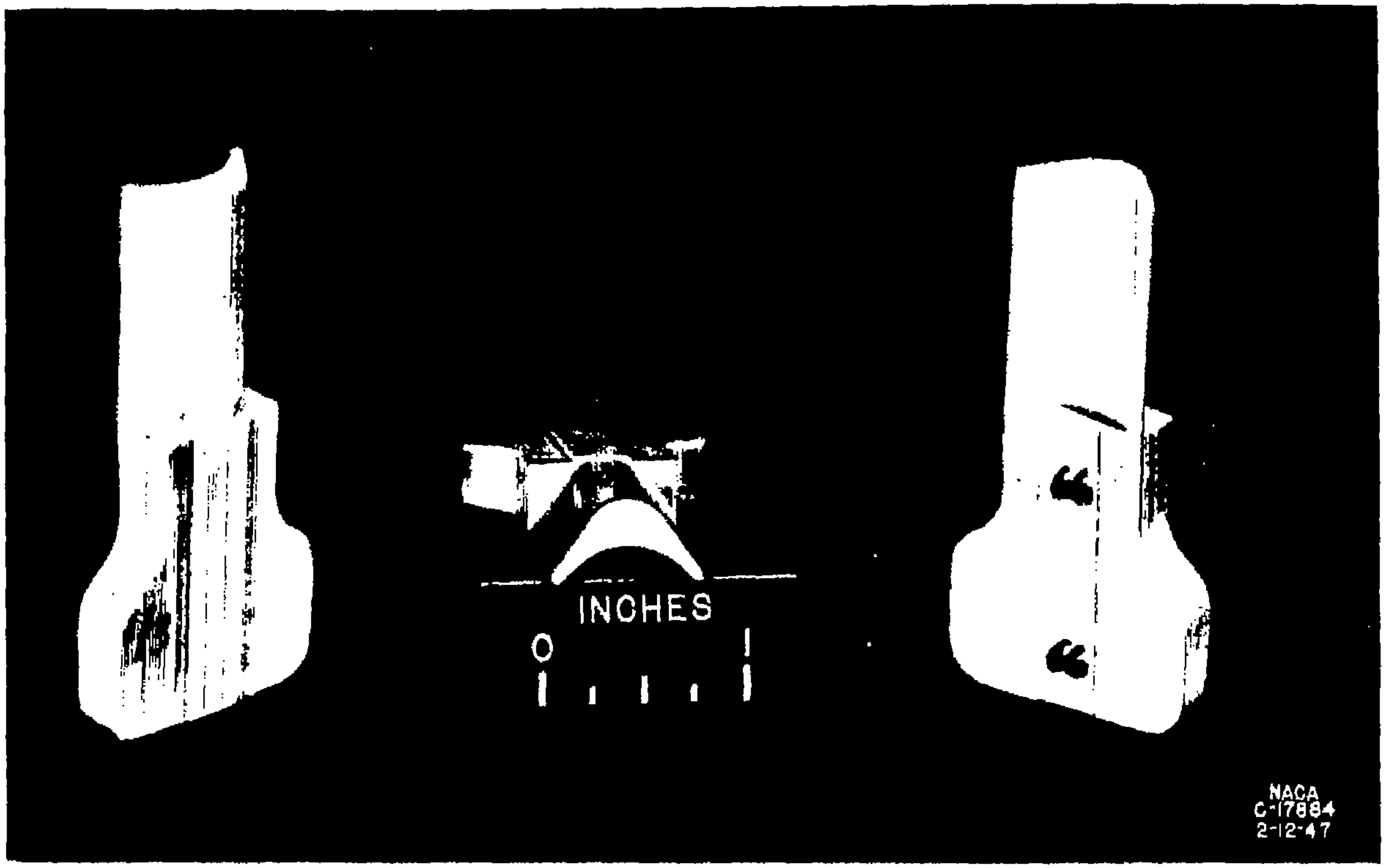

Figure 1. - First-design blades.

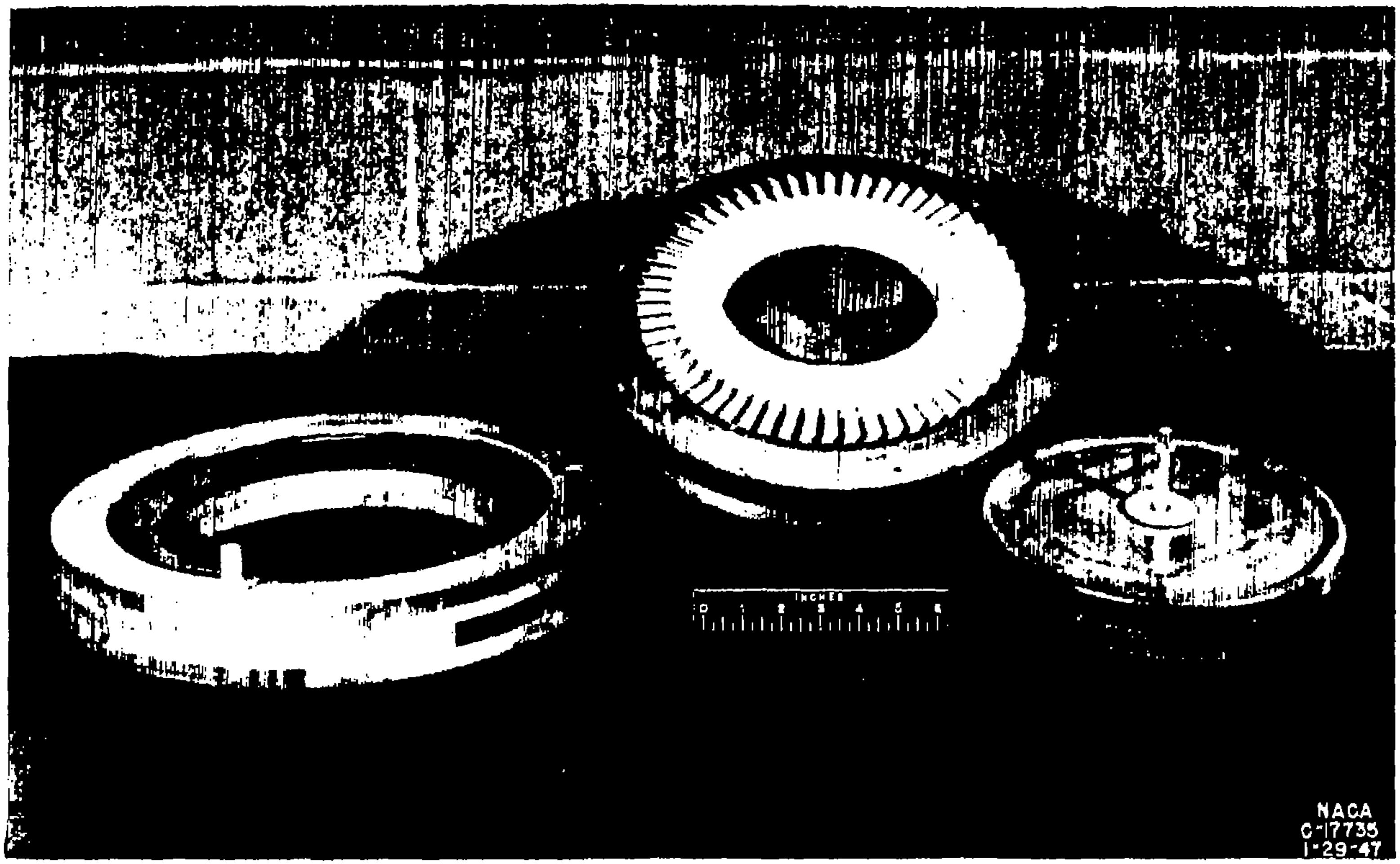

Figure 2. - Assembly fixture for ceramic-blade ring.

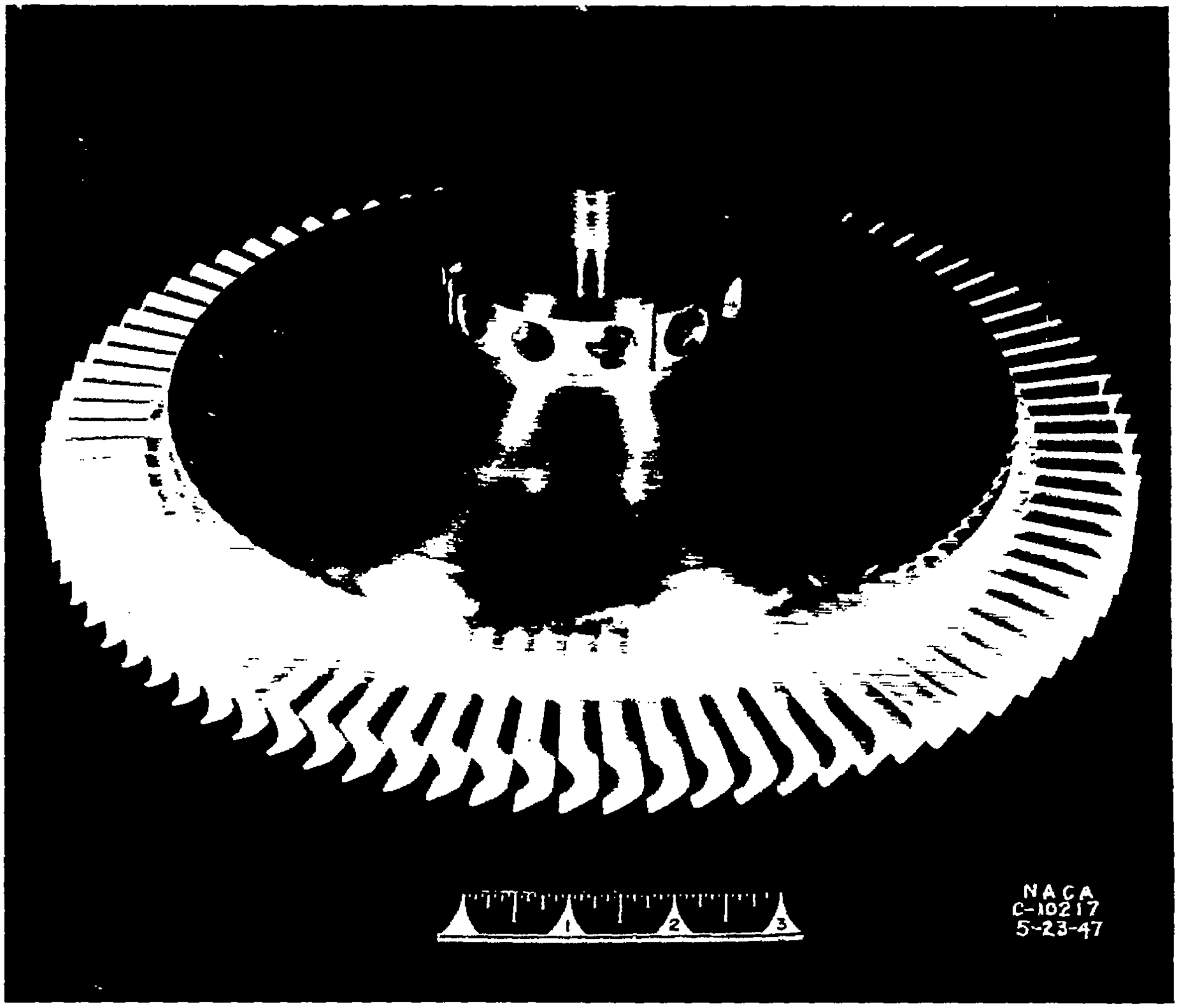

Figure 3. - Turbine-rotor assembly with first-design blades.

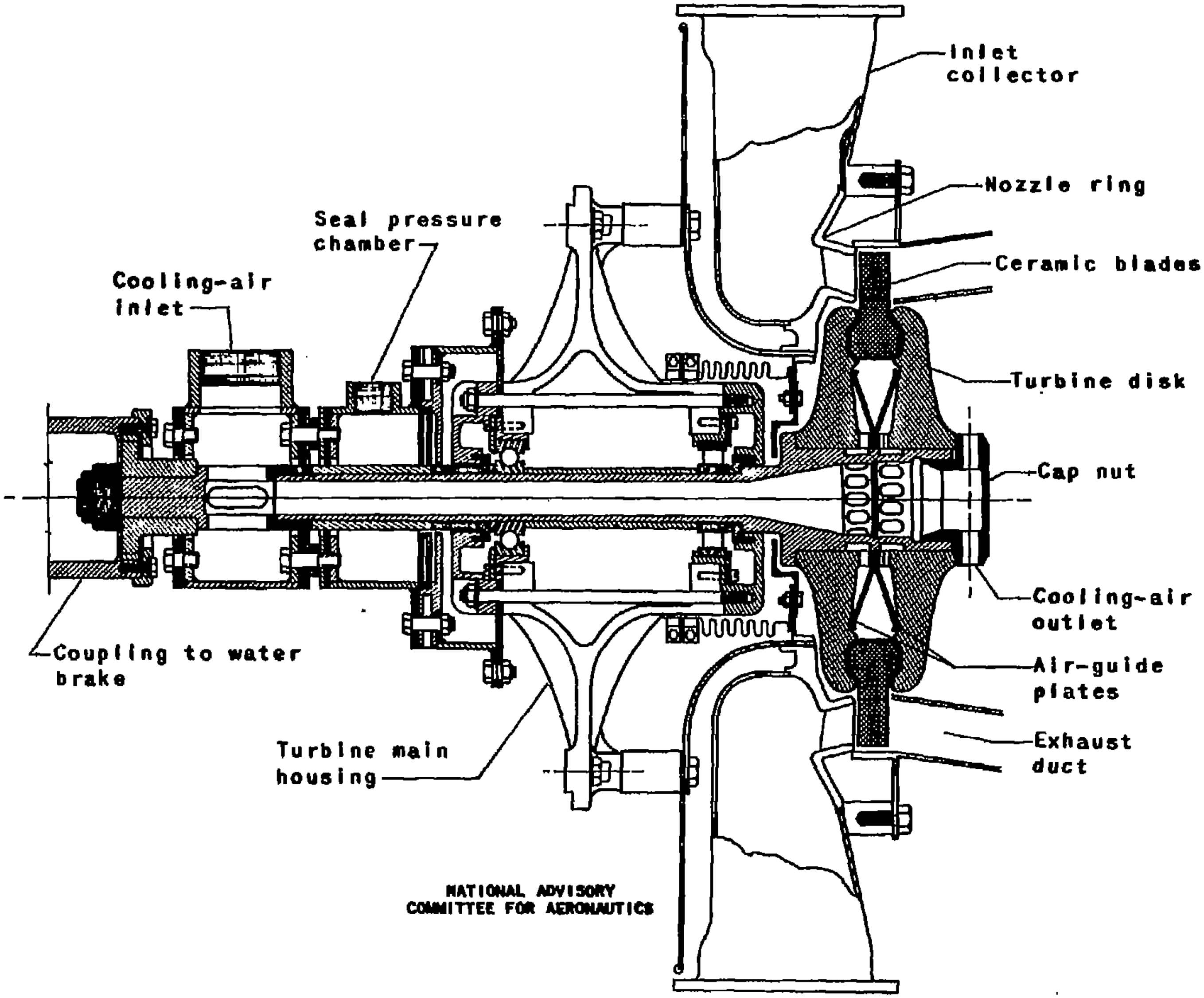

Figure 4. - Section through ceramic-blade gas turbine.

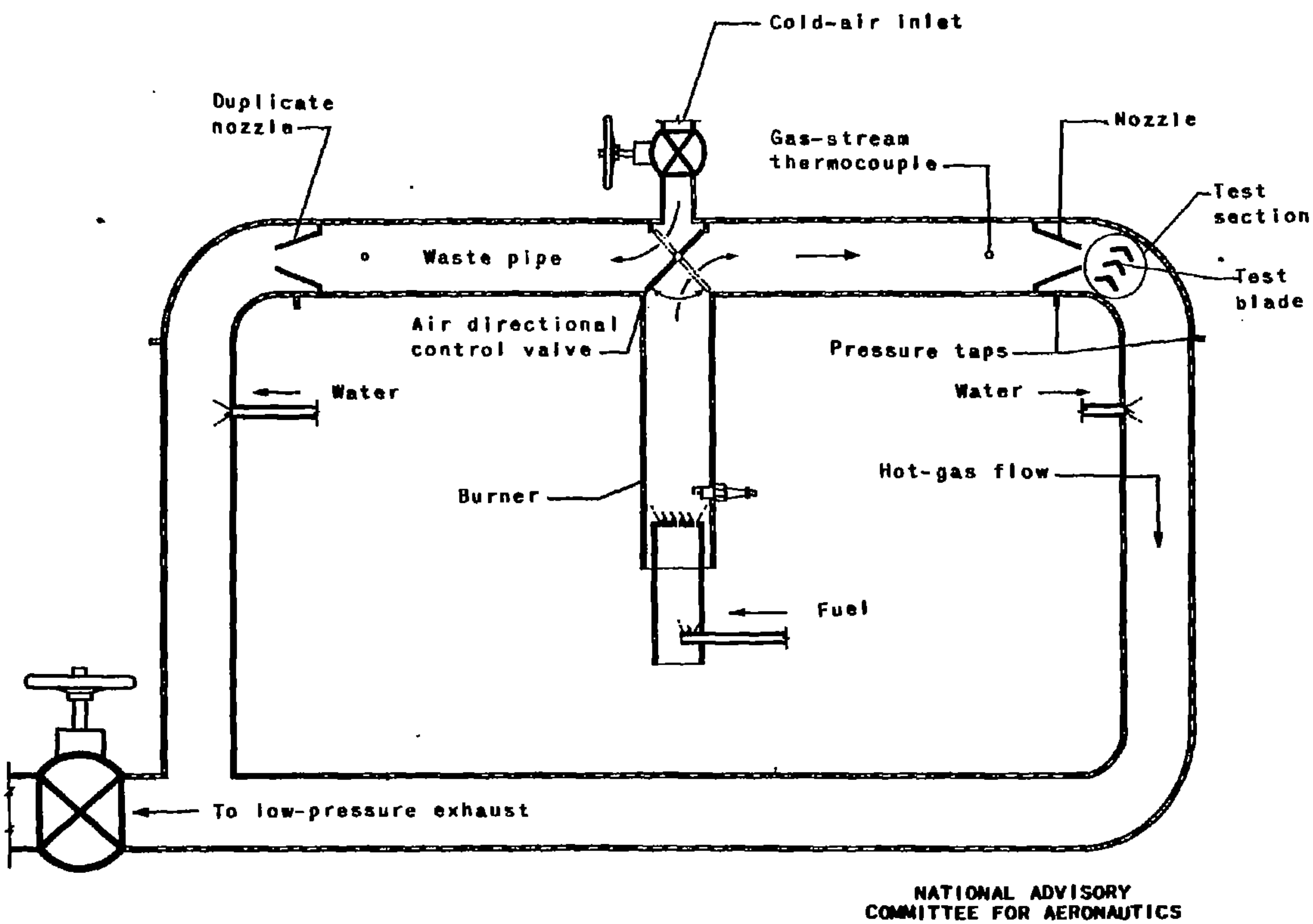

Figure 5. - Heat-shock test rig.

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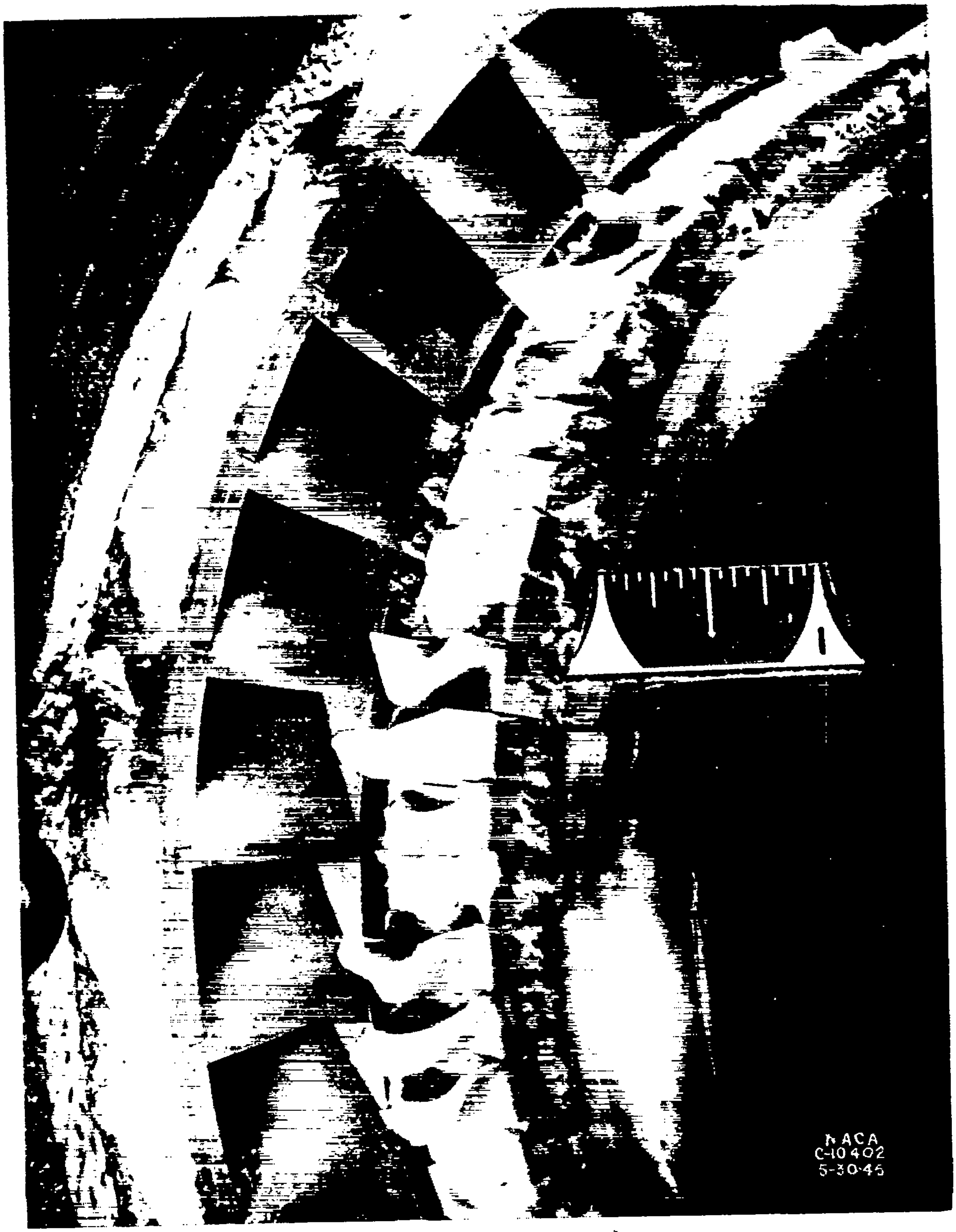

Figure 6. - Failure of first-design blades after $1\frac{1}{2}$ hours total running time.

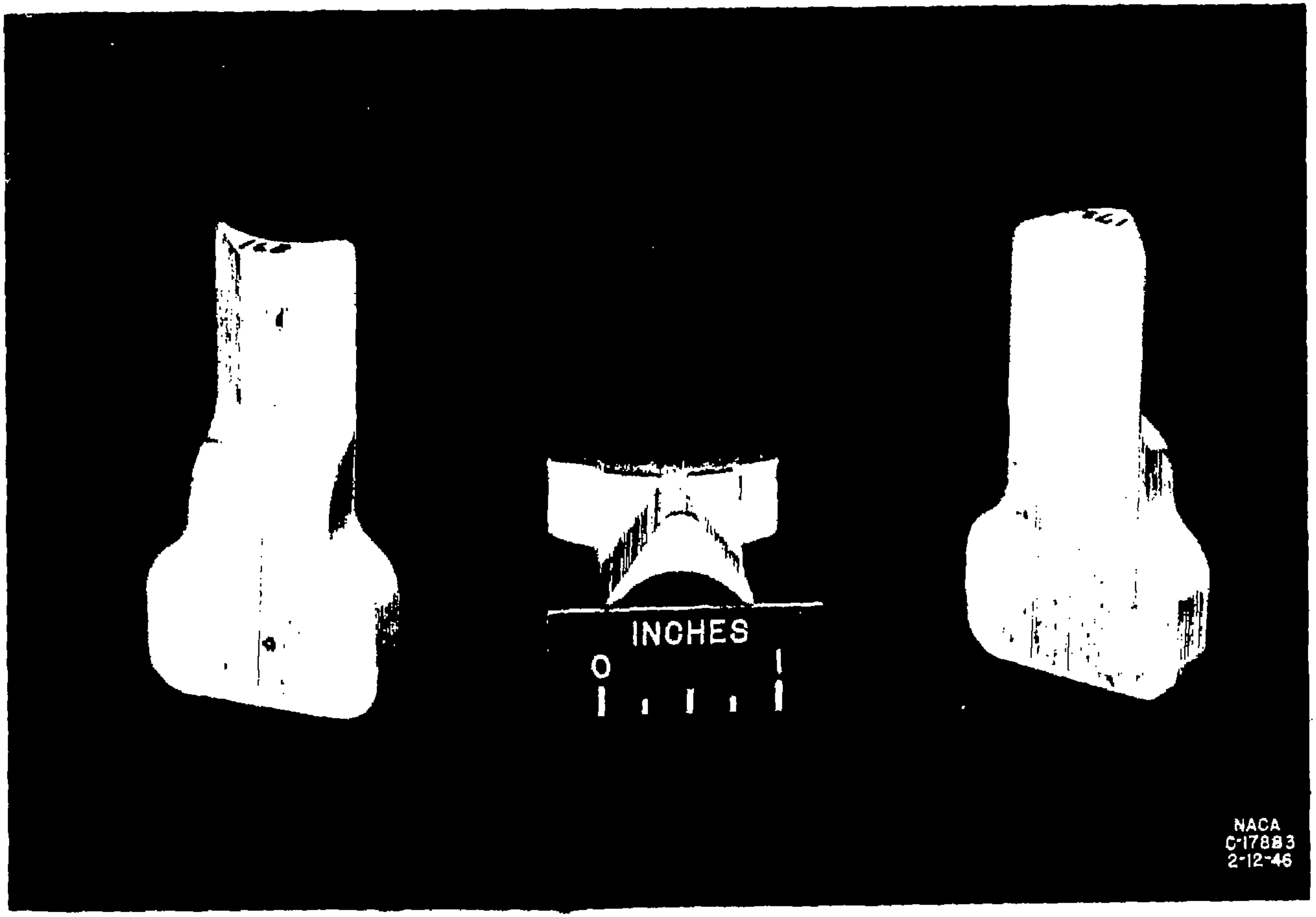

Figure 7. - Second-design blades.

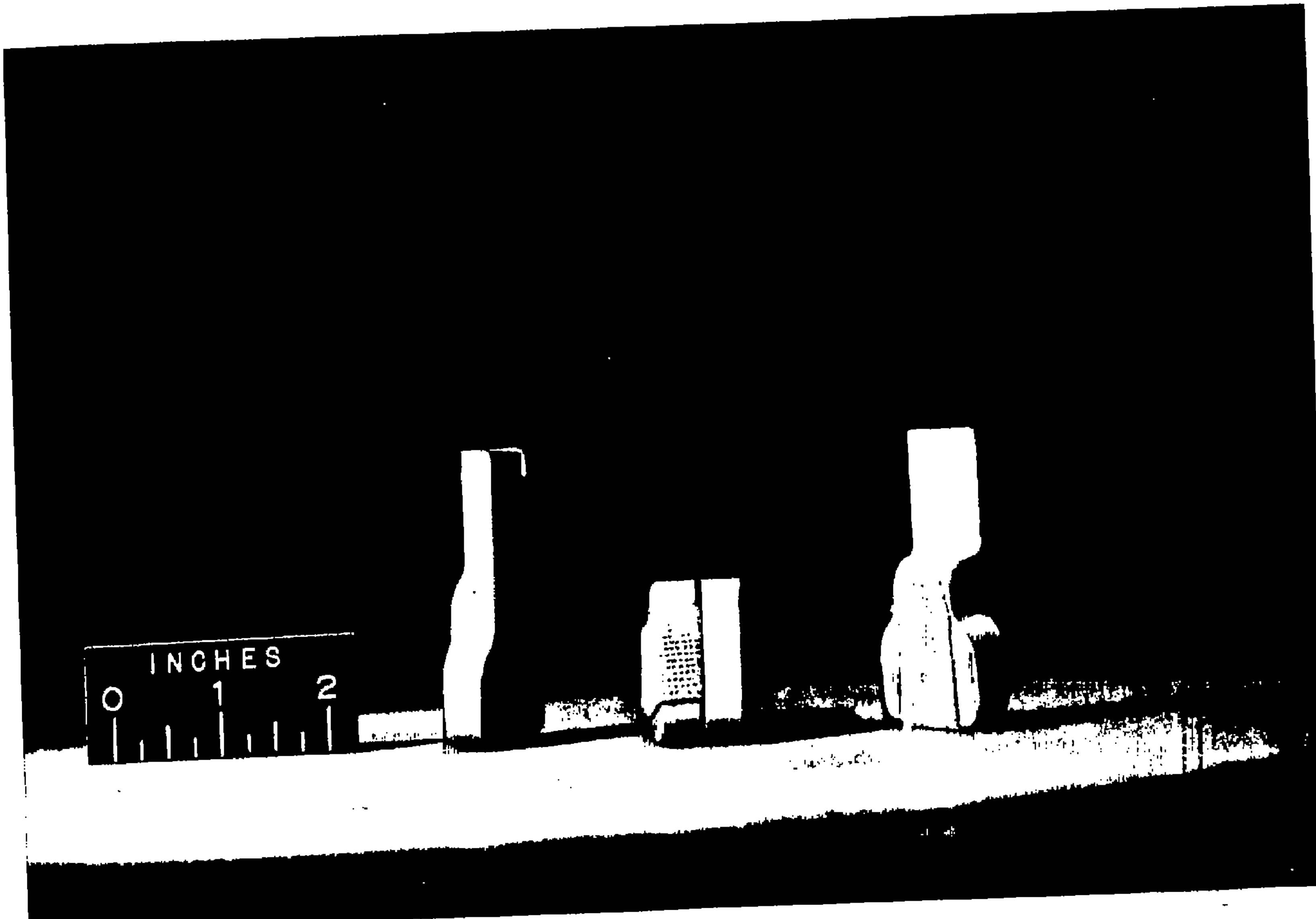

Figure 8. - Second-design blades showing gasket used on alternate blades.

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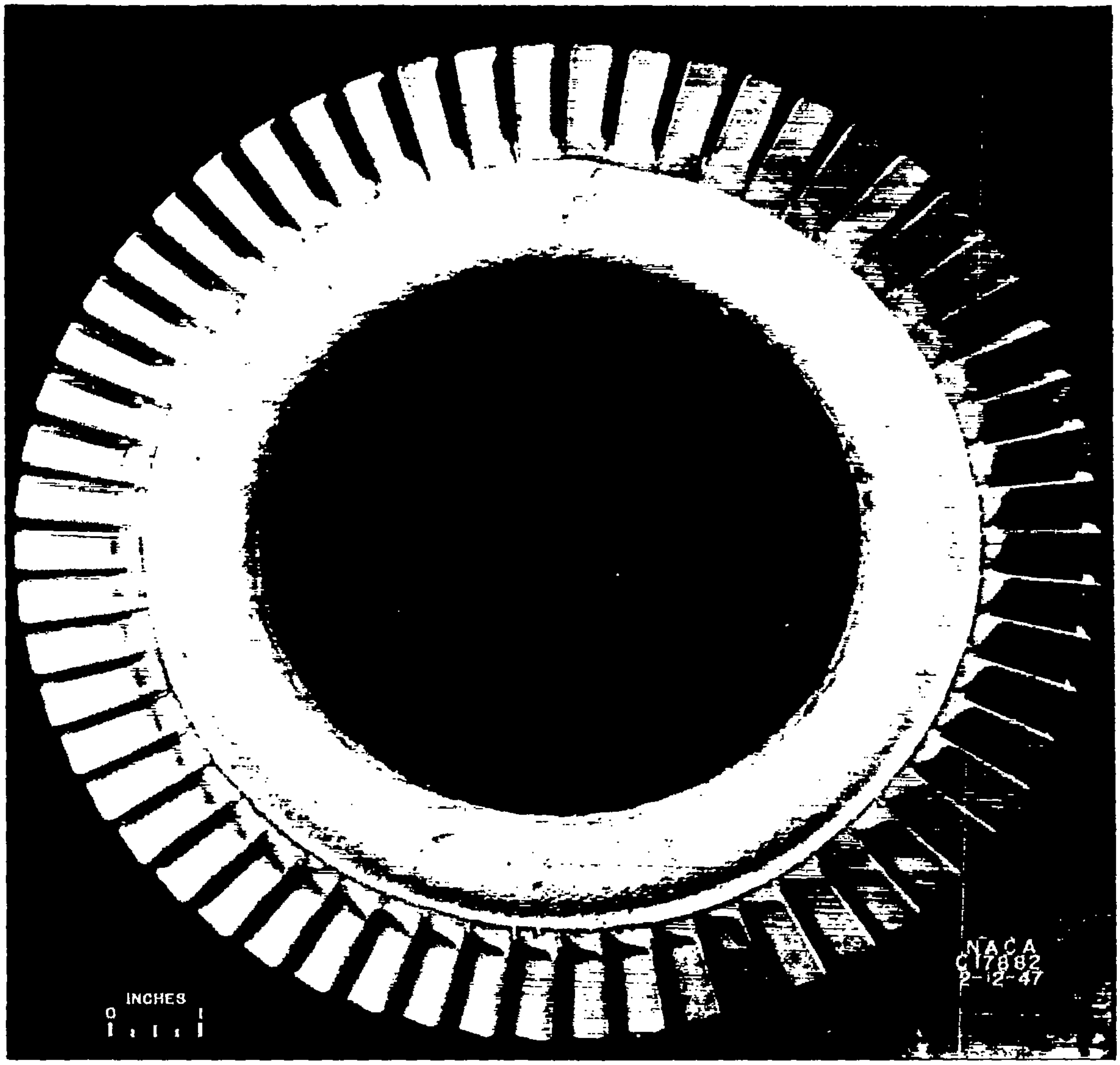

Figure 9. - Ceramic-blade ring with supporting gasket.

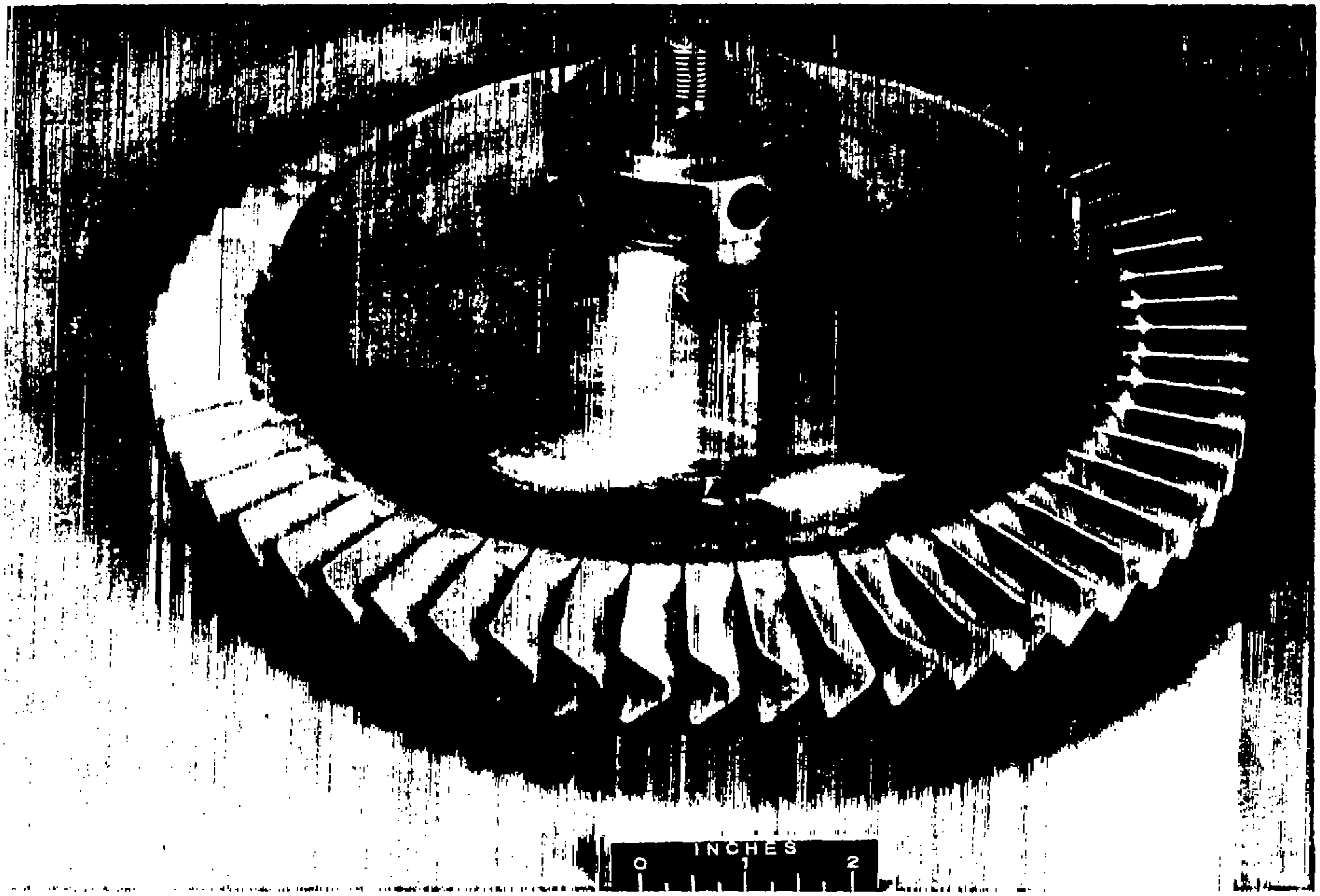

Figure 10. - Turbine-rotor assembly with second-design blades.

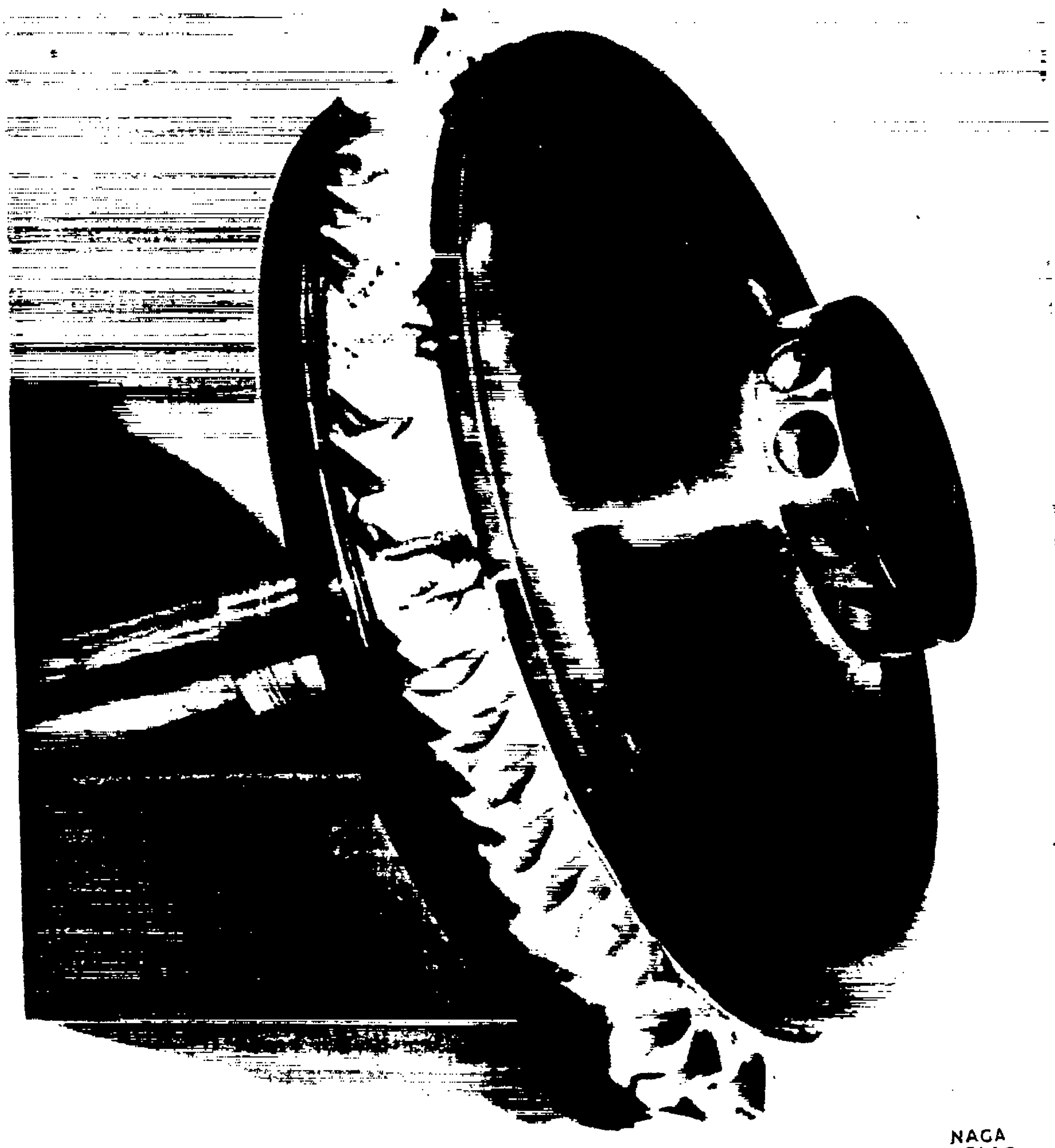

Figure 11. - Failure of second-design blades after 38 hours total running time.